THROUGH SCIENCE **WE DEFEND**

TECHNICAL REPORT
NATICK/TR-00/009

AD__________________

POTENTIAL USE OF THERMOELECTRIC MODULES FOR A SMALL QUANTITY THERMOELECTRIC BEVERAGE COOLER

by
Nathan Smith

November 1999

Final Report
May 1999 - September 1999

Approved for Public Release; Distribution is Unlimited

U.S. Army Soldier and Biological Chemical Command
Soldier Systems Center
Natick, Massachusetts 01760-5018

DISCLAIMERS

The findings contained in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

Citation of trade names in this report does not constitute an official endorsement or approval of the use of such items.

DESTRUCTION NOTICE

For Classified Documents:

Follow the procedures in DoD 5200.22-M, Industrial Security Manual, Section II-19 or DoD 5200.1-R, Information Security Program Regulation, Chapter IX.

For Unclassified/Limited Distribution Documents:

Destroy by any method that prevents disclosure of contents or reconstruction of the document.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	November 1999	FINAL May 1999 - September 1999

4. TITLE AND SUBTITLE
POTENTIAL USE OF THERMOELECTRIC MODULES
FOR A SMALL QUANTITY THERMOELECTRIC BEVERAGE COOLER

5. FUNDING NUMBERS
C 810CAF3131

6. AUTHOR(S)
Nathan Smith

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
U.S. Army Soldier and Biological Chemical Command
Soldier Systems Center
ATTN: AMSSB-RCF-E (N)
Natick, MA 01760-5018

8. PERFORMING ORGANIZATION REPORT NUMBER
NATICK/TR-00/009

9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES)

10. SPONSORING / MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION / AVAILABILITY STATEMENT
Approved for Public Release; Distribution Unlimited

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

This report explores the use of thermoelectric modules as a concept for a small quantity beverage cooler that would enable a soldier to cool a standard military canteen of water 30 degrees Fahrenheit in 30 minutes.

14. SUBJECT TERMS
THERMOELECTRIC MODULES, CANTEENS, LIGHTWEIGHT, COOLING, FIELD EQUIPMENT, BEVERAGES, PORTABLE, WATER, DISMOUNTED SOLDIER, WATER COOLING, PROTOTYPE

15. NUMBER OF PAGES
30

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	SAME AS REPORT

TABLE OF CONTENTS

LIST OF FIGURES

Figure

LIST OF TABLES

Table

PREFACE

This report explores the use of thermoelectric modules as a concept for a small quantity beverage cooler that would enable a soldier to cool a standard military canteen of water, 30 Fahrenheit degrees in 30 minutes. The work was performed by Nathan Smith of the Equipment & Energy Technologies Team, Combat Feeding Program, Soldier Systems Center, U.S. Soldier and Biological Chemical Command (SBCCOM), Natick, Massachusetts.

The citation of trade names in this report does not imply endorsement or other approval of such products.

POTENTIAL USE OF THERMOELECTRIC MODULES FOR A SMALL QUANTITY THERMOELECTRIC BEVERAGE COOLER

1. INTRODUCTION

The goal of the small quantity beverage cooler project is to develop a portable device by using a 24-Volt power source that is able to cool a standard military canteen of water, thirty Fahrenheit degrees in thirty minutes. This report explores the use of thermoelectric modules to cool water. During this part of the project a device was designed that used thermoelectric modules as a cooling device. Several configurations were designed and tested to develop the device resulting in a suggestion for a final product (yet to be produced).

2. BACKGROUND

The driving power of this thermoelectric canteen cooler is a thermoelectric module. The module acts as a heat pump and works based on the Peltier Effect. The Peltier Effect is when a temperature differential across two dissimilar materials is created when a DC current is applied between them. Thermoelectric modules are typically made of two ceramic wafers sandwiching P and N doped bismuth-telluride electrodes. As electrons move between the P and N poles, they also change energy states, creating a temperature differential between the two ceramic wafers. In effect, the module pumps heat from the cold side of the module to the hot side of the module. This allows heat to be extracted from a system attached to the cold side of the module, while adding heat to a system attached to the hot side of the module. The thermoelectric canteen cooler will remove heat from water in a canteen, and rejects heat to the environment.

Thermoelectric modules have specific power requirements to allow optimum performance. The applied voltage, Vmax, and current, Imax, specify the optimum power input. These variables are not the maximum allowable voltage and current, but the voltage and current at that result in the maximum temperature differential. If the module is underpowered, the delta T will be smaller. If the module is overpowered, heat dissipation within the module will prevent it from reaching the maximum delta T. The variable Q_{max} specifies the thermal load at which the module maintains a delta T equal to zero.

3. COMPONENTS

A simple prototype was built to test the effectiveness of the thermoelectric modules. The prototype consisted of four primary parts: a thermoelectric module, a heat sink, a water probe, and an aluminum block to link the module and probe. Thermal grease is required to assemble an effective prototype. It has a high thermal conductivity and allows for good heat transfer when applied between two surfaces. All part interfaces were coated with thermal grease. Assembling the prototype requires sandwiching the thermoelectric module between the heat sink and aluminum block. The block is secured and clamped by bolting it into the heat sink. The probe can then be inserted into the hole in the block. Several different components were evaluated to determine the effect on cooling performance.

Two heat sinks were evaluated. The gray heat sink, obtained from the Tellurex Corporation's Thermoelectric Cooler Prototype Starter Kit, is 3-inch by 5-inch extruded aluminum heat sink with twelve ¾-inch fins. This heat sink also has a ½-inch section without fins down its center for bolts to be inserted. The black heat sink was taken from a prototype built by Coolworks, Inc., and is 4-inches by 4-inches with sixteen 1.25-inch fins. Each sink came with a fan to increase the convective heat transfer to the air. The black heat sink's fan was slightly larger.

Two different thermal greases, also known as heat sink compounds, were evaluated. The first, supplied by Omega Engineering, Inc., was Omegatherm® "201" High Temperature High Thermal Conductivity Paste, which has a thermal conductivity of 2.5 W/m°K. It is gray in color with a thick consistency. The second thermal grease, supplied by McMaster-Carr Supply Co., was Chemplex 1381 Heat Sink Silicone. The Chemplex Silicone is white, and much less viscous than the Omegatherm®. It has a thermal conductivity of 0.75 W/m°K.

Two probes were evaluated. The first was a solid aluminum rod, while the other was an ammonia-charged heat pipe. The ammonia heat pipe was obtained from a chemical heater developed by Mainstream Engineering, Inc.

4. CALCULATIONS

The optimum material for the block was determined by calculating its volumetric heat capacity. A volumetric measure of heat capacity was used because the amount of material is dictated by the spatial arrangement. A low volumetric heat capacity is ideal because less energy is required to cool the material. Steel, aluminum, and copper; have volumetric heat capacities of 3866 kJ/m^{3}°K, 2439 kJ/m^{3}°K, 3439 kJ/m^{3}°K, respectively. Aluminum is best because it is readily available, has the lowest heat capacity, and is the lightest of the three materials.

The specifications of the Tellurex thermoelectric module number CZ1-1.4-127-1.14 and material properties of aluminum were used to create a mathematical model of a simple thermoelectric cooling probe. The probe was broken into forty-eight 1/8-inch long segments. An energy balance equation applied to each segment modeled the heat transfer to its surroundings of both convection to water and conduction to other segments. The general form of the equation, which was iterated for each segment of time, is as follows:

$$mc_p \frac{dT_n}{dt} = -kA_c \left.\frac{dT}{dx}\right|_{n,n-1} + kA_c \left.\frac{dT}{dx}\right|_{n+1,n} + hA_s(T_w - T_n) \qquad \textbf{\textit{Equation 1}}$$

where $m =$ mass

$c_p =$ heat capacity of aluminum

$T_n =$ temperature of segment n

$t =$ time
$k =$ thermal conductivity of aluminum
$A_c =$ cross-sectional area
$x =$ distance along probe
$h =$ convection coefficient
$A_s =$ surface area of each segment
$T_w =$ water temperature.

The first term on the right hand side of Equation 1 was replaced with Q_m, the heat drawn by the thermoelectric module, for the segment of the rod touching the block or the module. The equation for the segment at the end of the rod omits the second term on the right side of Equation 1. To determine a heat transfer for the whole rod the equation was integrated over the length of the rod and as time iterations were performed in Microsoft Excel. The convection coefficient was determined with the appropriate heat transfer correlation using fluid dynamics on a flat plate:

$$Nu = \left[0.825 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.492/\mathrm{Pr})^{9/16}\right]^{8/27}}\right]^2 \qquad \textbf{\textit{Equation 2}}$$

$$Nu = \frac{hL}{k} \qquad \textbf{\textit{Equation 3}}$$

The model solution was based on several assumptions that provided initial conditions. The hot side temperature was assumed to be 10° F above ambient. It assumed that the heat sink could dissipate as much heat as the module rejected from the hot side. The model also assumed no temperature stratification in the canteen water except for a transient dT close to the surface of the probe to produce convection estimations.

The model showed poor conduction along the probe resulting in the end of the probe remaining cold with no heat transfer from the water. This may have been the result of a high estimation of the convective heat transfer to each segment.

5. TEST SETUP AND PROCEDURES

All temperature data was obtained using the Fluke data logger and type K thermocouples in the Burner Test Facility. Unless otherwise noted, the thermocouples were covered in thermal grease and then secured to a surface with aluminum tape. The only exceptions occurred when the surfaces were obstructed. In these instances the thermocouple was secured against the surface, but not taped. Nine temperatures were recorded for both the block-cooling tests and the water-cooling tests. Both series of tests recorded the temperature of ambient air, the hot side of the module, the cold side of the module, and the tip of the heat sink fin. The hot side temperature was obtained by putting the thermocouple through a hole in the heat sink to the module surface. The cold side temperature was obtained by attaching the thermocouple to a small hold drilled in the block before the prototype was assembled. The hole was situated such that the thermocouple would be sandwiched between the edge of the module and the block. Three temperatures on the base of the heat sink, and two block surface temperatures were also recorded in the block-cooling tests. One block surface temperature, one heat sink base temperature, two probe temperatures, and the water temperature were recorded in the water-cooling tests. **Figures 1, 2, and 3** show the experimental set-up, including the prototype, fluke data logger, and module power source.

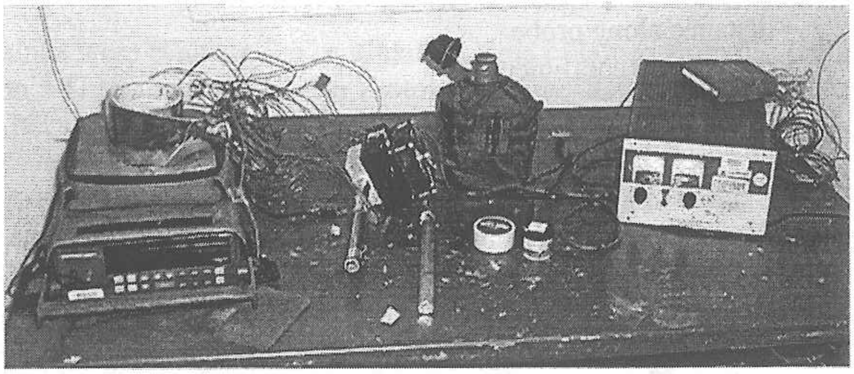

Figure 1. Complete Experimental Set-up.

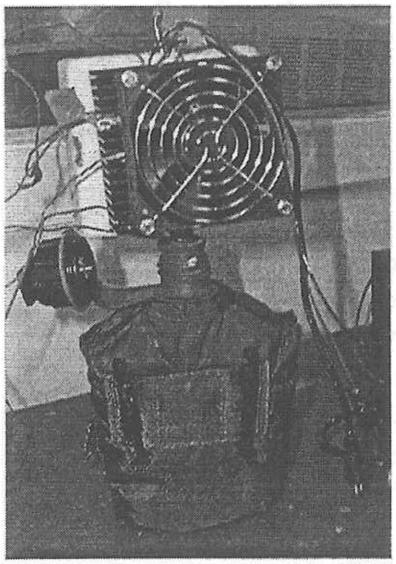

Figure 2. Prototype in Canteen.

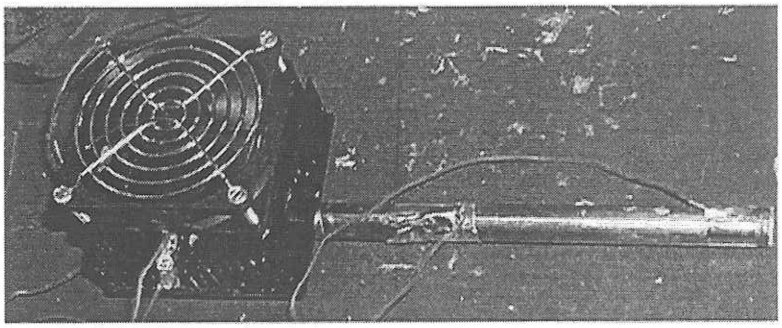

Figure 3. Prototype with Heat Pipe and Black Heat Sink.

6. RESULTS

6.1 BLOCK COOLING TESTS

Temperature data was recorded in the following tests for the thermoelectric cooling of the aluminum block that acts as the interface between the thermoelectric module and the canteen probe. The canteen probe was purposely left out of these tests so that it would not influence preliminary testing of various components. This allowed evaluation of two heat sinks, two thermal greases, and the power supply. All tests were run using Tellurex thermoelectric module number CZ1-1.4-127-1.14. All modules were powered with Lambda Electronics 10 Amp power supply (model number LK 343 FM). A 12-Volt DC Power Station III Converter powered the fan.

6.1.1 Omegatherm®, Black Heat Sink

This test was run to establish a baseline to compare different components of the prototype thermoelectric canteen cooler. The prototype used the black heat sink and fan; thermal interface between the components was provided with Omegatherm® "201" High Temperature High Thermal Conductivity Paste. The module was powered at 16.1 and 8 amps, listed in the module specifications to provide maximum performance. The results are shown in **Figure 4**. This prototype was not able to cool the block 30° F. It reached a steady state temperature of 43° F (6.1° C), which is only twenty-eight degrees below ambient. This test resulted in a steady state delta T across the module of 85° F (47.2° C). The block remained approximately the same temperature as the cold side of the module, so the heat transfer between the block and module was adequate. The tip of the heat sink fins were 20°F (11.1°C) higher than ambient. Typical forced convection heat exchangers run at 10°F over ambient, although this number can vary widely, so 20° F is an acceptable temperature difference.

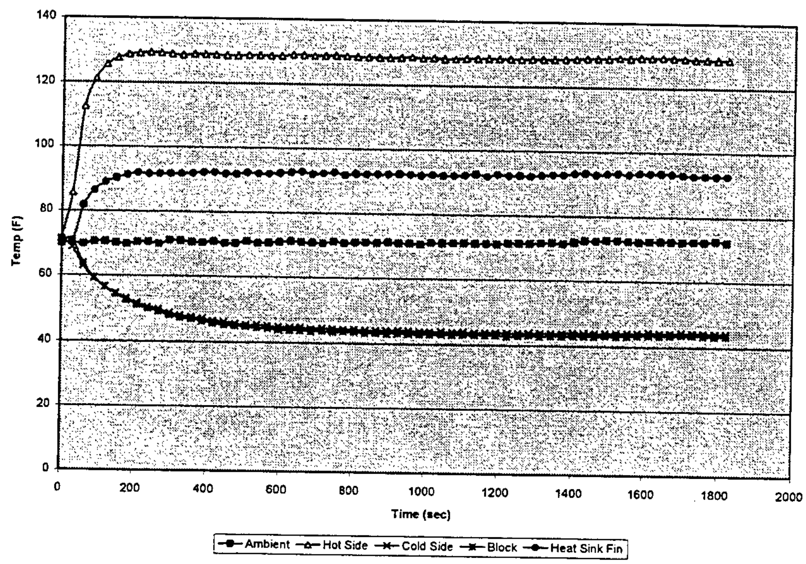

Figure 4. Test Results for Omegatherm and Black Heat Sink.

6.1.2 Omegatherm®, Gray Heat Sink

This test was run to compare the black and gray heat sink. The prototype used the gray heat sink and fan, and was put together using the Omegatherm® "201" High Temperature High Thermal Conductivity Paste. The module was powered at 16.1 Volts and 8 Amps. The results are shown in **Figure 5**. This prototype was not able to cool the block. In fact, after initially cooling the block 10° F, the block temperature rose nearly to ambient conditions. The steady state delta T across the module was 82° F (45.5° C). The block temperature profile followed that of the cold side of the module, indicating adequate heat transfer between the module and block. However, the system could not dissipate enough heat from the hot side of the module. The hot side module reached a temperature of 154° F (67.7° C). The heat sink fin temperature was 50° F (27.8° C) over ambient conditions; about five times the typical temperature differential. This indicates the gray heat sink is inadequate for the given thermal load.

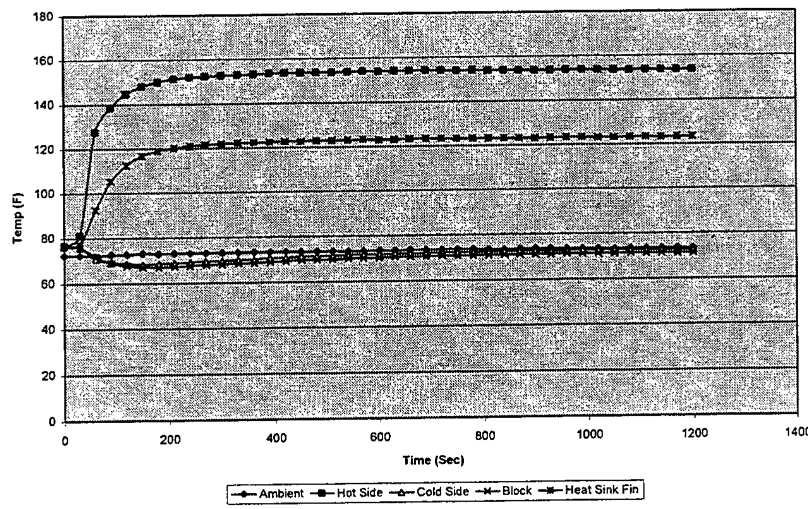

Figure 5. Test Results for Omegatherm and Gray Heat Sink.

6.1.3 Chemplex, Black Heat Sink

This test was run to compare Omegatherm® "201" High Temperature High Thermal Conductivity Paste and Chemplex 1381 Heat Sink Silicone. The prototype used the black heat sink and fan, and was put together using the Chemplex 1381 Heat Sink Silicone. The module was powered at 16.1 volts and 8 amps. The results are shown in **Figure 6**.

This prototype cooled the block 30° F in approximately four minutes. The block eventually reached a steady state temperature of 17° F (-8.3° C). The temperature differential across the module was 120° F (66.7° C). The block remained approximately the same temperature as the cold side of the module, indicating that the heat transfer between the block and module was sufficient. The tip of the heat sink fin was 32° F (17.8° C) higher than ambient. Although this temperature differential may be slightly larger than typical applications it is not an unreasonable condition.

6.1.4 Omegatherm®, Black Heat Sink (Retest)

This test was identical to test 6.1.1, except effects were made to reduce the thickness of the thermal grease the Omegatherm® "201" between the module and block/heat sink. This was done to try and determine why the higher rated thermal grease performed worse than the lower rated thermal grease. Because heat transfer depends inversely on film thickness, a thinner film would increase the thermal conductivity and explain the unexpected results. The thickness of the Chemplex and Omegatherm® assemblies were measured from the top of the heat sink to the base of the aluminum block, resulting in thicknesses of 2.586-inches and 2.592-inches, respectively. The Omegatherm® assembly

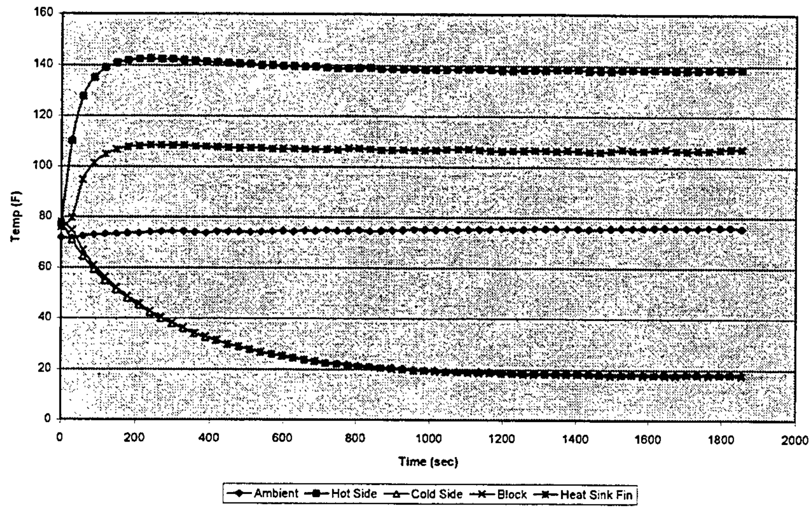

Figure 6. Test Results for Chemplex and Black Heat Sink.

was still six mils thicker than the Chemplex assembly. The module was powered at 16.1 volts and 8 Amps. The results are shown in **Figure** 7. This prototype cooled the block 30° F in approximately five minutes, with the block reaching a steady state temperature of 22°F (-5.6°C). This performance is almost as good as the Chemplex prototype. The temperature differential across the module was 100° F (55.6° C). The block remained approximately the same temperature as the cold side of the module, indicating that the heat transfer between the block and module was sufficient. The tip of the heat sink fin was 26° F (14.4° C) higher than ambient. This temperature differential may be slightly larger than that found in typical applications, but the system was able to cool the block fairly quickly.

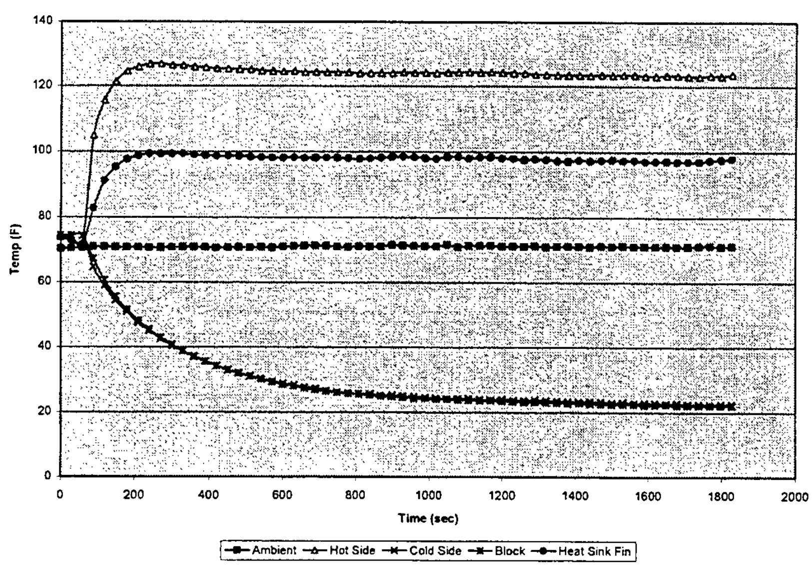

Figure 7. Test Results for Retest of Omegatherm and Black Heat Sink.

6.1.5 Two Modules

This test was run to evaluate the effect of surface area on the heat transfer between the module and the block/heat sink. The prototype used the black heat sink and fan, and was put together using the Chemplex 1381 Heat Sink Silicone. The modules were run in parallel, with each being powered at 10 Volts and 5 Amps, the maximum allowed by the power supply. It was assumed that the two modules would run with the same hot and cold side temperatures. Using two modules doubled the area of heat rejection to the heat sink. The block-module interface area increased by a factor of 1.575 because the length of the block was smaller than the length of two modules. The results are shown in **Figure 8**.

This prototype cooled the block 30° F in approximately 2.5 minutes, with the block reaching a steady state temperature of 15°F (-9.4°C). The temperature differential across the module was 102° F (56.7° C). The block remained approximately the same temperature as the cold side of the module, indicating that the heat transfer between the block and module was sufficient. The tip of the heat sink fin was 27° F (15° C) higher than ambient. This temperature differential may be slightly larger than that found in typical applications, but the system was able to cool the block extremely quickly.

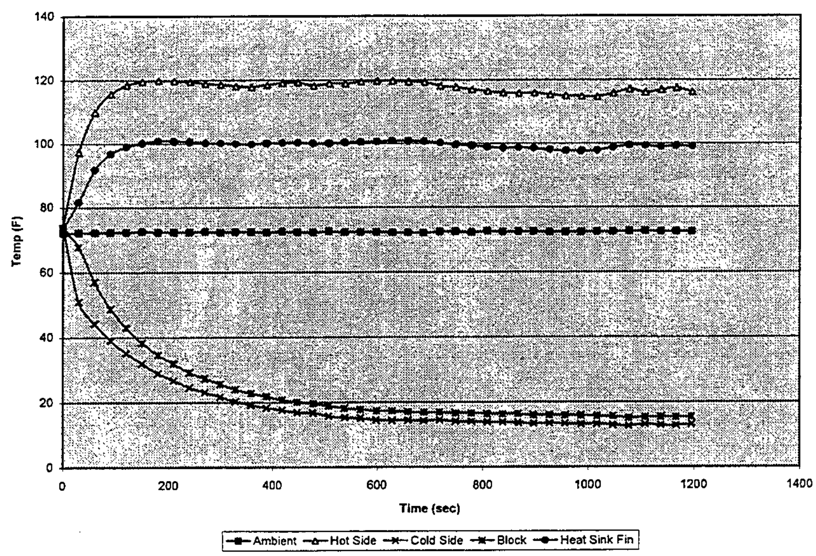

Figure 8. Results of the Two Module Test.

6.1.6 One Module at 10V

This test was run so that the cooling ability of the two module prototype could be evaluated. The single module consumed the same power as each module in the two module test. The prototype used the black heat sink and fan, and was put together using the Chemplex 1381 Heat Sink Silicone. The module was powered at 10 Volts and 5 Amps. V_{max} and I_{max} were not used so the results could be compared to the previous test of two modules. The results are shown in **Figure 9**.

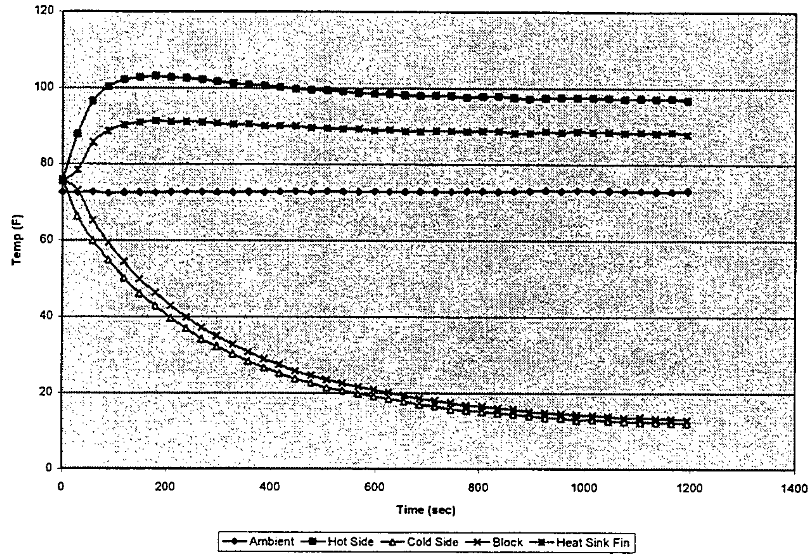

Figure 9. Test Results for One Module at 10 Volts.

This prototype cooled the block 30° F in approximately four minutes, with the block reaching a steady state temperature of 13°F (-10.6°C). The temperature differential across the module was 102° F (56.7° C). The block remained approximately the same temperature as the cold side of the module, indicating that the heat transfer between the block and module was sufficient. The tip of the heat sink fin was 27° F (15° C) higher than ambient. This temperature differential may be slightly larger than that found in typical applications, but the system was able to cool the block quickly.

6.1.7 One Module at 20V

This test was run so that the cooling ability of the module at a voltage closer to the output of military equipment could be evaluated. The prototype used the black heat sink and fan, and was put together using the Chemplex 1381 Heat Sink Silicone. The power supply could not output enough voltage so the module was powered at 20 Volts and 10 Amps, the maximum allowable by the power supply. The results are shown in **Figure 10.** This prototype could not cool the block 30° F. The block reached a minimum temperature of 49°F (9.4°C). The temperature differential across the module was 110° F (61.1° C). The block remained approximately the same temperature as the cold side of the module, indicating that the heat transfer between the block and module was sufficient. The tip of the heat sink fin was approximately 55° F (30.5° C) higher than ambient.

6.2 WATER COOLING TESTS

After the optimum components were selected based on results of the block-cooling tests, the experimental setup was expanded by including the probe inserted into the block for the cooling of water. The water-cooling tests used the optimum components as determined from the block cooling tests. These components are the Chemplex Silicone, black heat sink, V_{max} and I_{max}, and one module. The tests also had a probe inserted into the aluminum block to transfer heat from the water to the block and thermoelectric module.

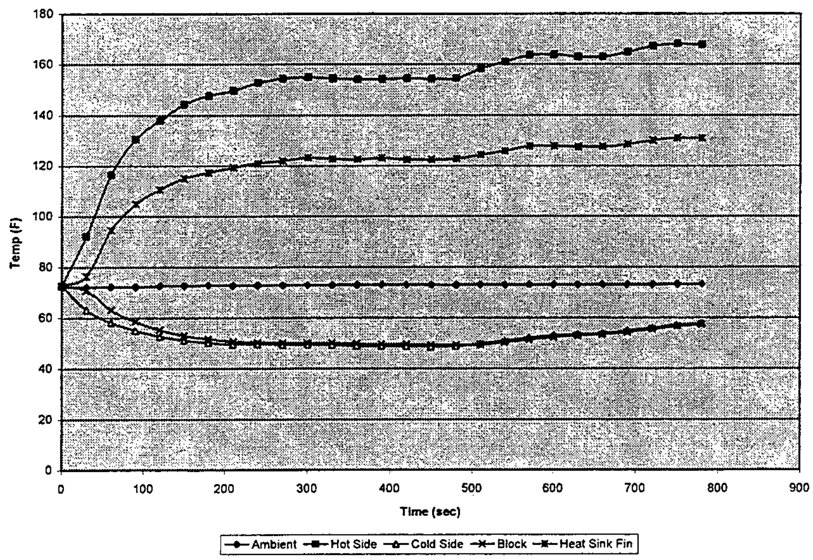

Figure 10. Results for One Module at 20 Volts.

6.2.1 Heat Pipe

This prototype, using a steel-ammonia heat pipe probe, could only cool the water 7° F (3.8° C) in thirty minutes. The increased thermal load on the cold side of the module limited the cold side temperature to 53° F (11.6° C). There was a 2° F temperature rise to the block, and then another 2° F rise to the probe. There was also an 8° F temperature increase from the top of the probe to the bottom of the probe. These temperature differentials indicate barriers to optimum system performance. The steady state temperature difference across the module was approximately 86° F (47.8° C). The additional thermal load increased the heat sink/ambient temperature differential to 29° F (16.1° C), which may be a little too high for optimum performance. The results are shown in **Figures 11 and 12**.

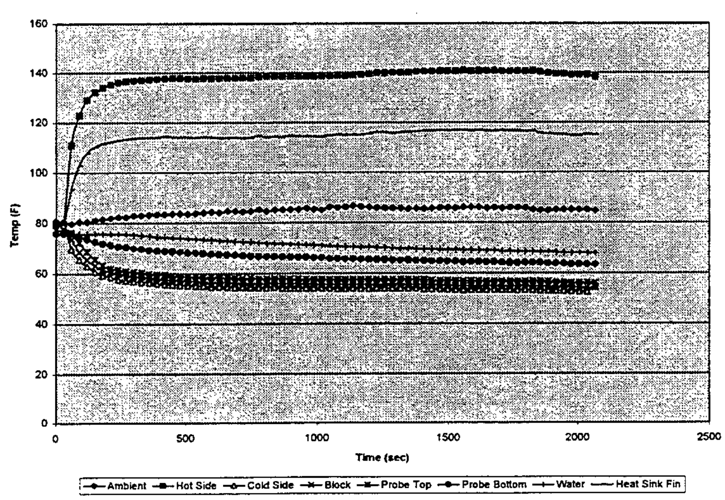

Figure 11. Test Results of Cooling Water with an Ammonia Heat Pipe.

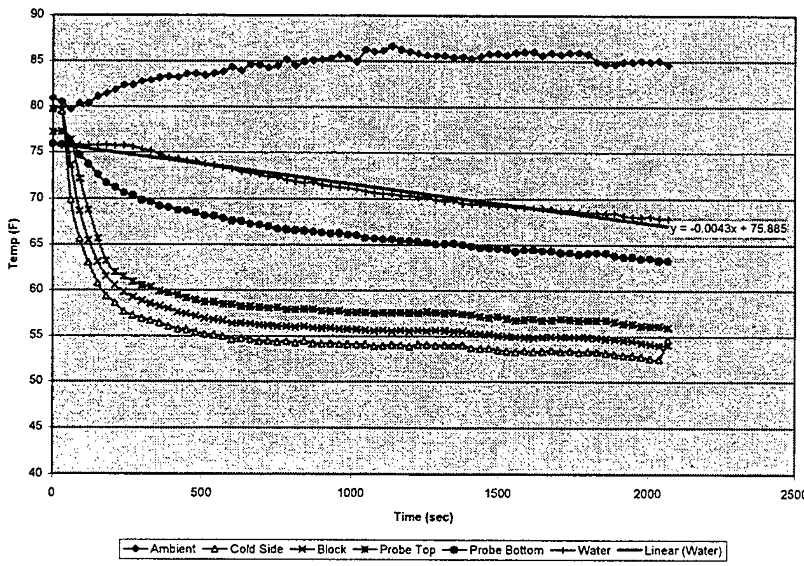

Figure 12. Test Results of Cooling Water with an Ammonia Heat Pipe.

6.2.2 Aluminum Pipe

This prototype, using a solid cylindrical aluminum probe, could only cool the water 2° F (1.1° C) in thirty minutes. The increased thermal load on the cold side of the module limited the cold side temperature to 46° F (7.7° C). There was a 2° F temperature rise to the block, and then another 12° F rise to the probe. There was also an 11° F temperature increase from the top of the probe to the bottom of the probe. All these temperature differentials indicate losses in the system. The steady state temperature difference across the module was approximately 91° F (32.8° C). The heat sink fin/ambient temperature differential was 26° F (14.4° C), which is not an unreasonable value. The results are shown in **Figure 13 and 14.**

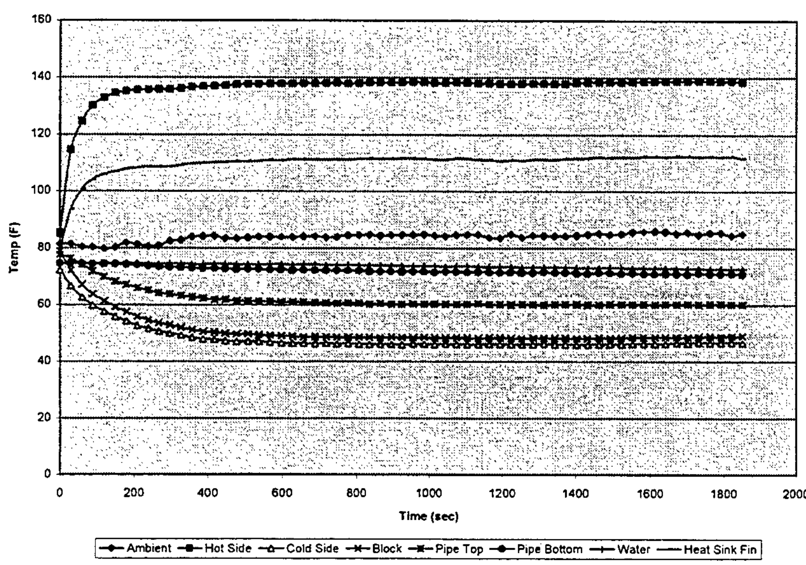

Figure 13. Test Results for Cooling Water with an Aluminum Probe.

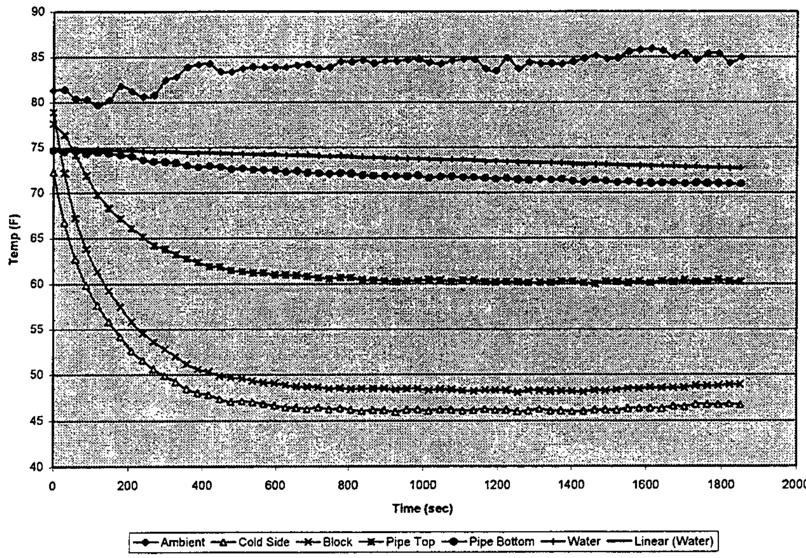

Figure 14. Test Results for Cooling Water with an Aluminum Probe.

6.2.3 Heat Pipe Test

To determine if the heat pipe was limiting the heat transfer, a constant cooling source was applied to one end of the heat pipe, while the other end was immersed in a canteen of water. The results of this test are shown in **Figure 15**. The bumps in the plot of pipe temperatures occurred when ice was added to the ice bath. However, the bath can be assumed to be 32° F (0° C). Despite the ice bath, the heat pipe only reached a minimum temperature of approximately 50° F (10° C), while remaining mostly in the range of 55° F to 65° F. This is a 30° F temperature differential, which indicates the heat along the length of the pipe was inadequate.

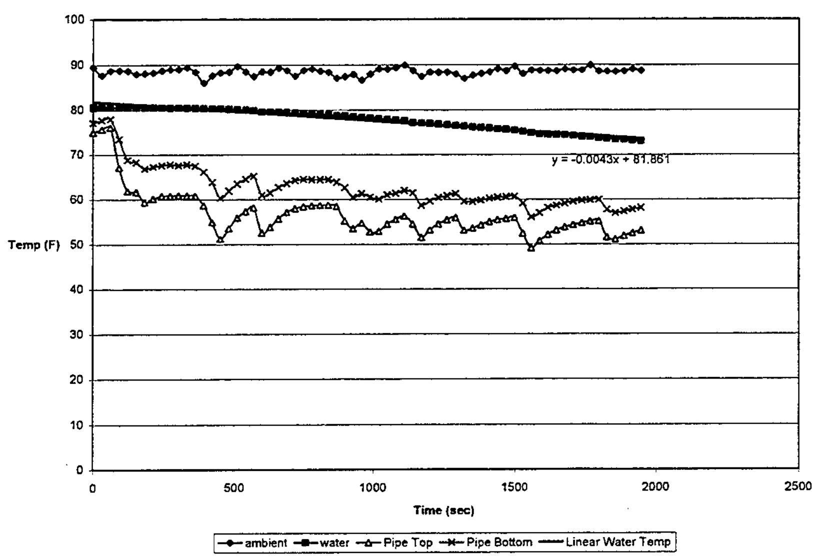

Figure 15. Test Results Heat Pipe Analysis.

7. ANALYSIS

7.1 EVALUATON OF PROTOTYPE COMPONENTS

7.1.1 Heat Sink and Fan

Figures 16, 17, and 18 plot the performance of the heat sinks. The black heat sink performed better in all regards. The prototype with the black heat sink was able to cool the block, while the gray was not. Better heat transfer occurred between the black heat sink and the module than between the gray heat sink and the module, as evidenced by the lower hot side temperature of the black heat sink. Finally, the temperature differential between ambient and the fin tip of the black heat sink was approximately 30°F smaller than that of the gray heat sink.

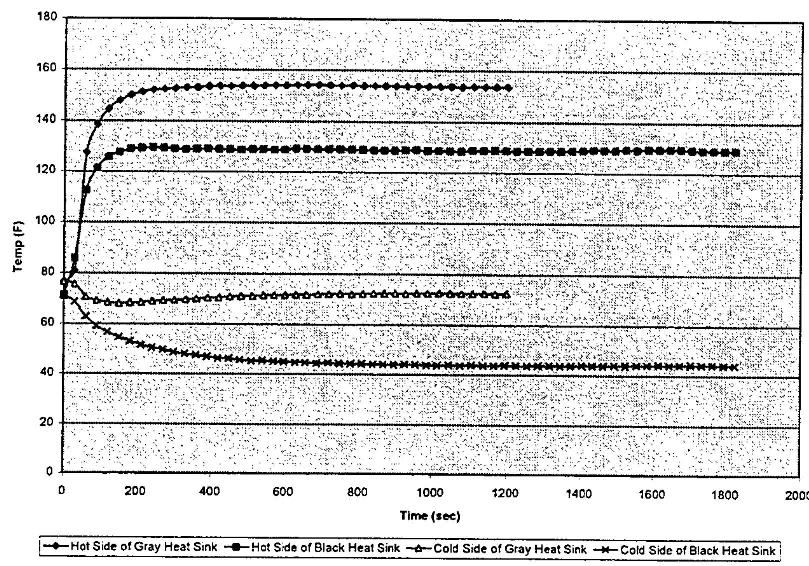

Figure 16. Heat Sink Performance by Module Temperature.

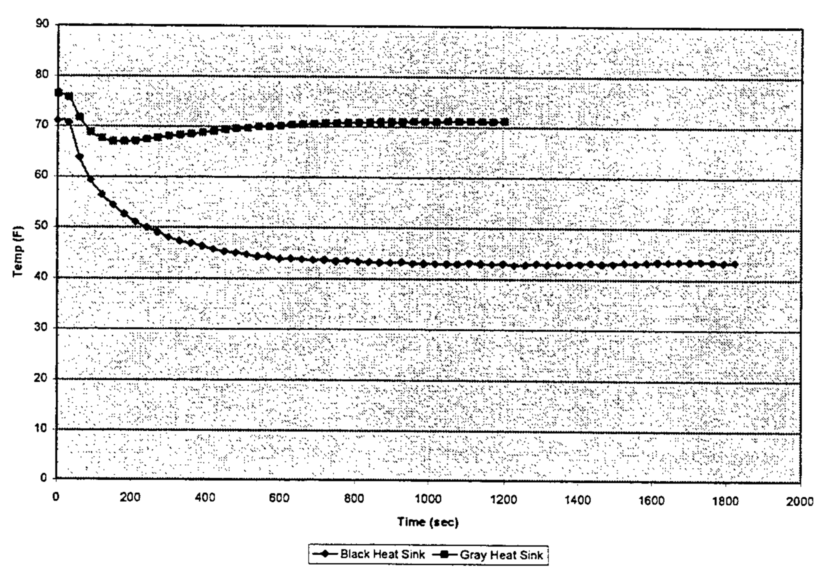

Figure 17. Heat Sink Performance by Block Temperature.

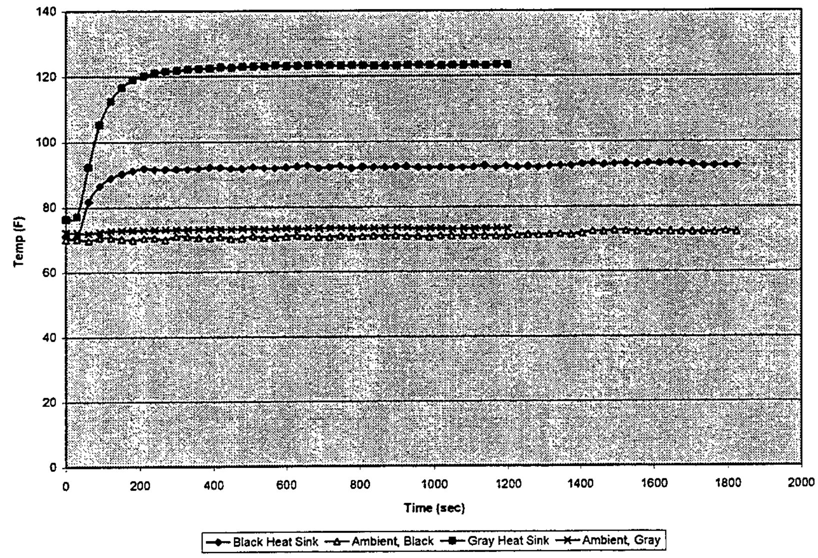

Figure 18. Heat Sink Performance by Heat Sink Fin Temperature.

7.1.2 Thermal Grease

The Chemplex Silicone outperformed the Omegatherm® Paste despite its lower thermal conductivity. **Figures 19 and 20** show plots of the hot and cold side module temperatures, and the temperature of the block. A second test of the Omegatherm® paste was conducted to see if the superiority of the Chemplex silicone resulted from its lower viscosity. Extreme care was taken to make the layer of Omegatherm® paste as thin as possible. Limiting the film thickness improved the cooling ability of Omegatherm® paste nearly to that of Chemplex silicone. It is suspected that the lower viscosity of the Chemplex silicone ensured that all of the surface area available for heat transfer was utilized due to the fact that upon compression of the assembly, it spread over the entire surface. In addition to superior performance, the Chemplex Silicone was easier to apply than the Omegatherm® Paste.

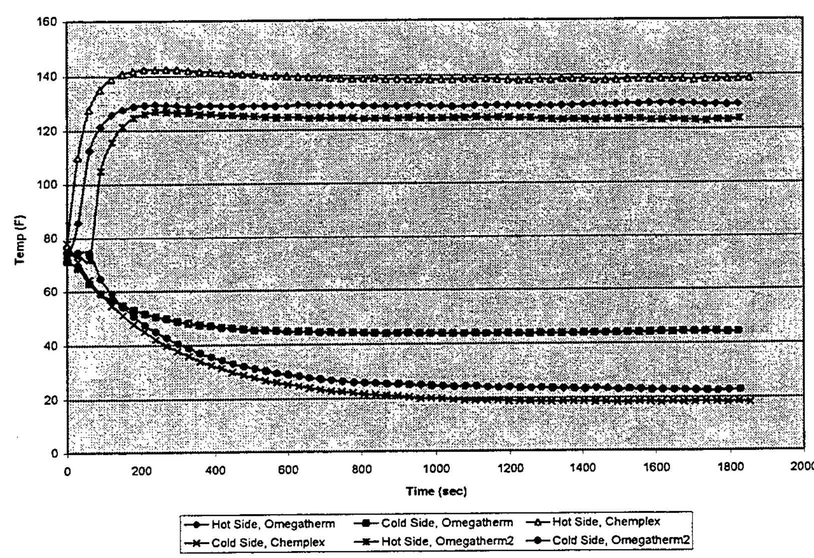

Figure 19. Thermal Grease Performance by Module Temperature.

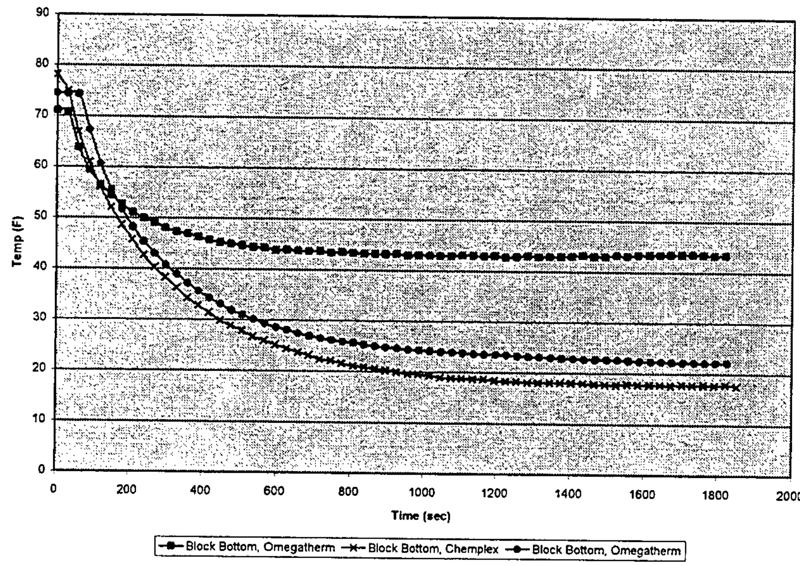

Figure 20. Thermal Grease Performance by Block Temperature

7.1.3 Number of Modules

Although the optimum voltage and current could not be applied, changing the number of thermoelectric modules in the prototype, and thus the surface area available for heat transfer from the module improved cooling ability. **Figures 21 and 22** compare the block temperature and module temperature of the one and two module prototypes. The power required for two modules is less than that for one module, with a slight increase in performance.

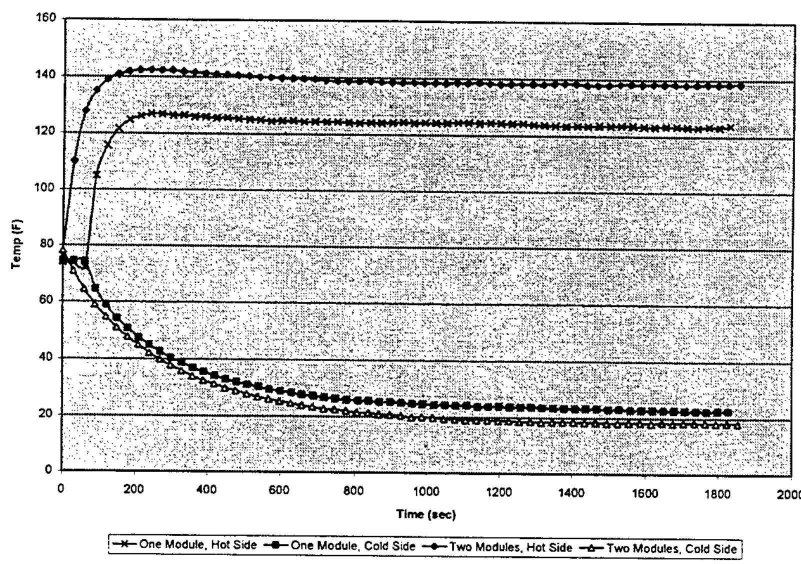

Figure 21. Performance of One and Two Module Prototypes by Module Temperature.

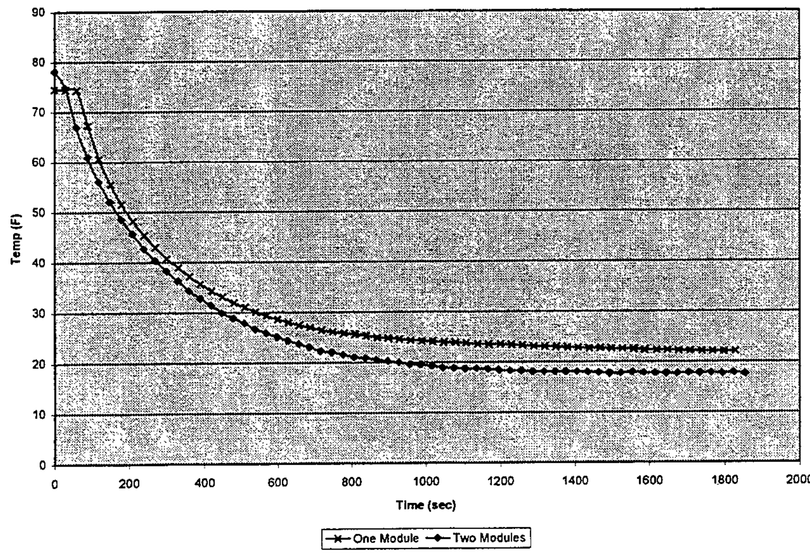

Figure 22. Performance of One and Two Module Prototypes by Block Temperature.

7.1.4 Power

Figures 23 and 24 plot the performance of the thermoelectric module at different powers. Increasing the power beyond the values of V_{max} and I_{max} is detrimental to prototype performance. The hot side temperature becomes too high because of overheating. This causes the cold side to be too high, limiting the heat transfer between the module and block. There was very little difference in performance at 16 Volts and 10 Volts. The slightly better performance of the 10 Volt prototype was probably due to the smaller temperature difference across the module, and the heat sink being better suited to a smaller load. Currently if the HMMWV 24 Volt power supply is used on a module optimized for 16 volts, the performance of the module will suffer.

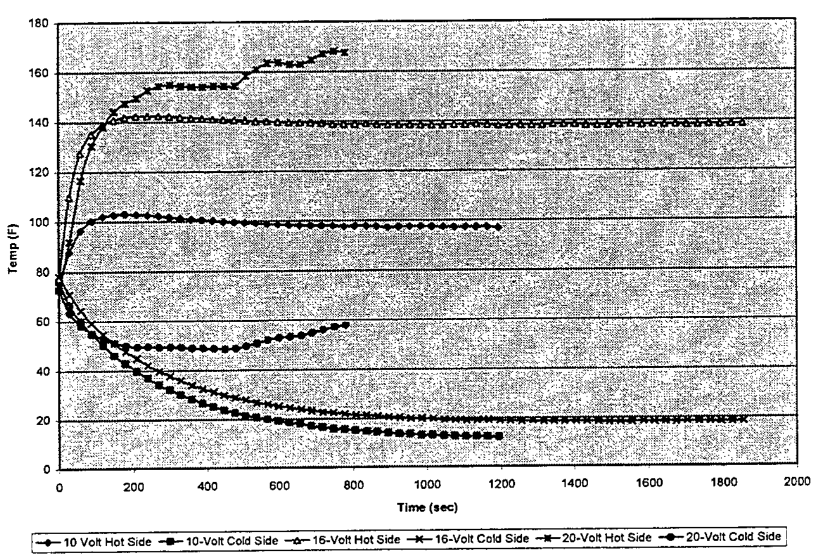

Figure 23. Comparison of Performance at Different Voltages by Module Temperature.

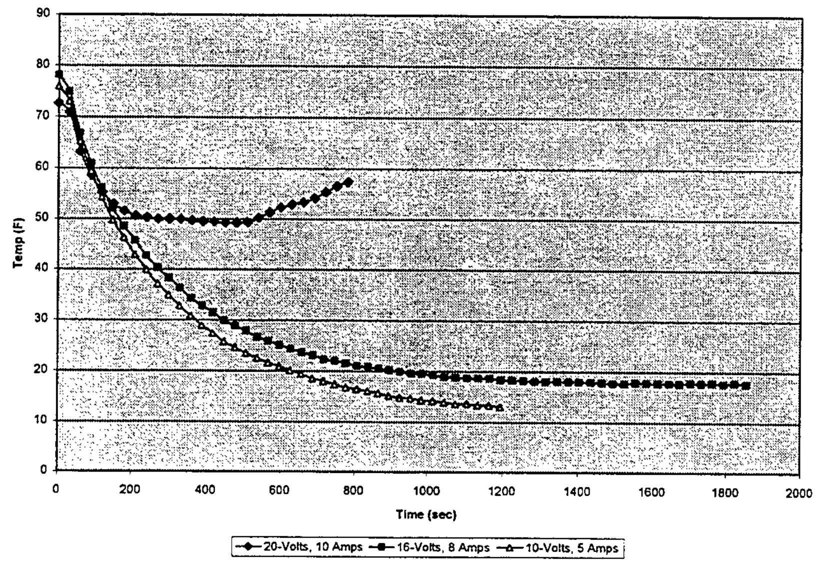

Figure 24. Comparison of Performance at Different Voltages by Block Temperature.

8. CONCLUSIONS

There are several aspects of the prototype design that could be changed to improve performance. These include eliminating unnecessary interfaces that obstruct heat flow, increasing the area on which heat transfer from the system takes place, increasing the convection heat transfer between the water and probe, the probe material, and optimizing module and power supply interface.

8.1 Unnecessary Interfaces

One of the most obvious problems with the current prototype design is the presence of extra material interfaces that obstruct heat flow, specifically between the probe and the block. Experimental data shows that there is a 1.8° F (1° C) temperature difference between the pipe and the block. The thermal conductivity of the Chemplex Grease is 0.75 W/m°K and the interface surface area is 5.89 in^2 (0.0038 m^2). The pipe thickness and block hole diameter were measured to be 0.752 inches and 0.755 inches, respectively. This results in a film thickness of three mils (0.0000762 m). Therefore, since

$$Q = kA\frac{dT}{dx}$$ ***Equation 4***

where x is the film thickness, the heat transfer across silicone film is 37.4 W. If the heat pipe were integrated into the block, the limiting conductivity would not be the interface, but that of the aluminum itself. For comparison, the thermal conductivity of aluminum is 237 W/m°K.

One possible solution to this problem would be to combine the heat pipe and block into a single unit. It could be manufactured a couple of different ways. Thin walled aluminum tubing could be welded or brazed onto an aluminum block with a hole the same size as the internal diameter of the tube. This would create a component the same size and shape as the probe and block combination. Other methods of manufacture include casting, or machining from one piece of metal. The welding method is probably the best for prototype building, although casting is probably best for mass production.

8.2 Probe Material and Fluid

The heat pipe cycle consists of the successive evaporation and condensation of a working fluid. These experiments began the heat pipe cycle by extracting heat from the ammonia vapor, condensing the ammonia on the internal walls of the pipe where they fall to the base of the pipe. The condensation of ammonia causes a pressure drop in the pipe, which results in evaporation of liquid. The evaporation of liquid results in energy being absorbed from the surroundings. The continuous movement of fluid results in faster heat transfer along the pipe. The heat pipe used in this experiment was obtained from Mainstream Engineering, Inc., and was originally intended for use in a chemical canteen cooler. The pipe was charged with ammonia, which has very good properties for some heat pipe applications, one of which is its high heat of vaporization. Ammonia's high heat of vaporization may actually limit the heat transfer because of the small amount of energy being extracted in the cycle.

Table 1. Heat of Vaporization of Refrigerants

Fluid	Heat of Vaporization @ 80° F
	(BTU/lb_m)
Ammonia	224.39
R-12	242.8
R-22	110.80
R-134a	89.3*

The poor test results could indicate limitations of ammonia in this temperature range. **Table 1** shows a list of potential refrigerant fluids. Because of its high heat of vaporization, ammonia can absorb a lot of heat, however, it also requires a large amount of heat to generate a vigorous evaporation-condensation cycle. One application of ammonia heat pipes in a heated bridge uses a propane gas-fired furnace to heat a mixture of propylene glycol and water. This antifreeze mixture circulates through a separate piping loop to evaporators, heating the ammonia in the heat pipes (http://www.vdot.state.va.us/info/hotbridge.html). Ammonia outperformed other refrigerants for this application, but the high temperature differential of the heat input required suggests ammonia may not be good for this small-scale application.

Typically in A/C dehumidification equipment, R-22 is used in heat pipes. These systems have a similar delta T. A fluid, such as R-22 or R-134a, with a lower heat of vaporization may improve heat transfer because more fluid would condense per amount of extracted energy. This increased speed and volume of fluid movement may increase heat transfer significantly.

8.3 Convection

Another possible limitation of the model is the convective heat transfer from the water to the heat pipe. Using the previously mentioned heat pipe experiment shown in **Figure 15**, the convective heat transfer coefficient can be calculated. Assuming the canteen to be perfectly insulated, the heat transfer from the water is given by the equation

$$Q = mc_p \frac{dT}{dt} \qquad \textit{\textbf{Equation 5}}$$

where Q is heat, m is the mass, c_p is the heat capacity of water and dT/dt is water temperature change rate. The resulting Q is 9.07 W (30.96 BTU/hr). The convection, $Q=hA\Delta T$, where h is the convective heat transfer coefficient, A is the surface area of the pipe, and ΔT is the temperature difference between the pipe and the water. Using the average pipe temperature to calculate ΔT, h equals 85.2 $W/m^{2\circ}K$ (0.104 $BTU/in^2hr^\circ F$). To obtain the required minimum 35 Watts of heat transfer, using equation 5, a temperature differential of 69.5° F (38.6° C) is required. This would cause the probe to form ice and thus limit the heat transfer. Therefore the temperature difference must be lower. If experimentation shows that this heat transfer rate cannot be obtained, a method of circulating the water could greatly increase the convective heat transfer.

8.4 Configuration and Number of Modules

The current configuration of thermoelectric modules results in two deficiencies. One is the limited surface area available for heat transfer. The second is the amount of material required to create the necessary surfaces to thermally link the modules to the heat pipe. The third is the optimal use of the available power. These deficiencies can be remedied by adjusting the configuration and number of thermoelectric modules. The configuration of modules is limited by the amount of material, while the number is limited by the power consumption.

Increasing the number of thermoelectric modules will increase the surface area at the region of heat transfer between the module and block or heat sink. However, the number of modules is limited by the power consumption. So, to increase the heat transfer area, several smaller modules with a combined Q_{max} larger than that for the single large module must be used. Several potential modules are listed in **Table 2**. Several small modules would also provide for optimal use of available power. The number of modules and their V_{max} and I_{max} characteristics and parallel or series wiring could maximize area and power.

Table 2. Small Thermoelectric Module Specifications.

Company	Model Number	I_{max} (amps)	Q_{max} (W)	V_{max} (volts)	dT_{max} (°C)	Power (W)	length (mm)	width (mm)	thickness (mm)
Melcor	CP 1.4-17-045L	8.5	9.2	2.06	67	17.51	15	15	3.3
Melcor	CP 1.4-31-045L	8.5	16.8	3.75	67	31.875	20	20	3.3
Melcor	CP 2-17-10L	9	10.3	2.06	70	18.54	22	22	5.6
Melcor	CP 2-17-06L	14	16	2.06	67	28.84	22	22	4.6
Americool	TM-71-1.0-3.0M	3	14.9	8.6	71	25.8	22.4	22.4	4
Ferro-Tec	6300/071/030	3	16	9.8	72	29.4	22.4	22.4	3.18
Ferro-Tec	6300/035/040	4	10.5	4.8	72	19.2	29.8	15.1	4.16
Ferro-Tec	6300/031/060	6	14	4.3	72	25.8	20	20	4.16
Ferro-Tec	6300/017/085	8.5	10.8	2.3	72	19.55	15.1	15.1	3.94
TECA	950-35	6	14.8	3.9	66	23.4	30.5	14.6	3.8

9. RECOMMENDATIONS

A second-generation prototype should be constructed to address the limitations identified in the previous section. The amount of material required to thermally link the heat pipe and the modules should be minimized. A cylinder would be the optimal shape from a material standpoint because it minimizes the ratio of surface area to volume. However, because the modules must attach to a flat surface, a multi-faced cylindrical shape, such as an octagon or hexagon, would be a potential solution. **Table 3** shows the potential configurations (hexagonal and octagonal) and the percent reduction in material and percent increase in Q_{max} from the original prototype configuration.

Table 3. Quantitative Improvements Resulting from the Use of Small Thermoelectric Modules.

Company	Model Number	Six Module Configuration				Eight Module Configuration			
		Power*6 (W)	Q_{max}*6 (W)	Volume Red. %	Q_{max} Increase %	Power*8 (W)	Q_{max}*8 (W)	Volume Red. %	Q_{max} Increase %
Melcor	CP 1.4-17-045L	105.06	55.2	90.0	-29.2	140.08	73.6	79.5	-5.6
Melcor	CP 1.4-31-045L	191.25	100.8	68.8	29.2	255	134.4	43.9	72.3
Melcor	CP 2-17-10L	111.24	61.8	56.3	-20.8	148.32	82.4	23.1	5.6
Melcor	CP 2-17-06L	173.04	96	56.3	23.1	230.72	128	23.1	64.1
Americool	TM-71-1.0-3.0M	154.8	89.4	53.4	14.6	206.4	119.2	18.4	52.8
Ferro-Tec	6300/071/030	176.4	96	53.4	23.1	235.2	128	18.4	64.1
Ferro-Tec	6300/035/040	115.2	63	79.7	-19.2	153.6	84	58.6	7.7
Ferro-Tec	6300/031/060	154.8	84	68.8	7.7	206.4	112	43.9	43.6
Ferro-Tec	6300/017/085	117.3	64.8	89.7	-16.9	156.4	86.4	79.0	10.8
TECA	950-35	140.4	88.8	81.6	13.8	187.2	118.4	61.3	51.8

One possible configuration is shown in **Figures 25, and 26**. A unified heat pipe/block should be constructed out of aluminum, utilizing the octagonal design. The part can be cast as one piece, or welded or brazed from two pieces, provided that the bond is strong enough to withstand the pressure exerted by the working fluid. Eight Teca (or some other candidate) modules, would be sandwiched between the octagonal face and heat exchangers. This design requires a custom-built heat exchanger because of the unique placement of the modules. One problem that must be overcome is providing enough clamping force to provide adequate contact between the modules and heat sink. This would be accomplished by cutting the heat sink into eight pieces, one for each module. Each section will have a small extension of the base at the top and bottom. Each extension will be threaded such that a tapered thread clamping mechanism can secure the heat sink sections to each other, and to the probe to provide the necessary clamping force. The top clamping mechanism must have tabs on which to mount the fan. Retaining rings on the circumference of the fins would add rigidity to the structure and prevent damage to the modules.

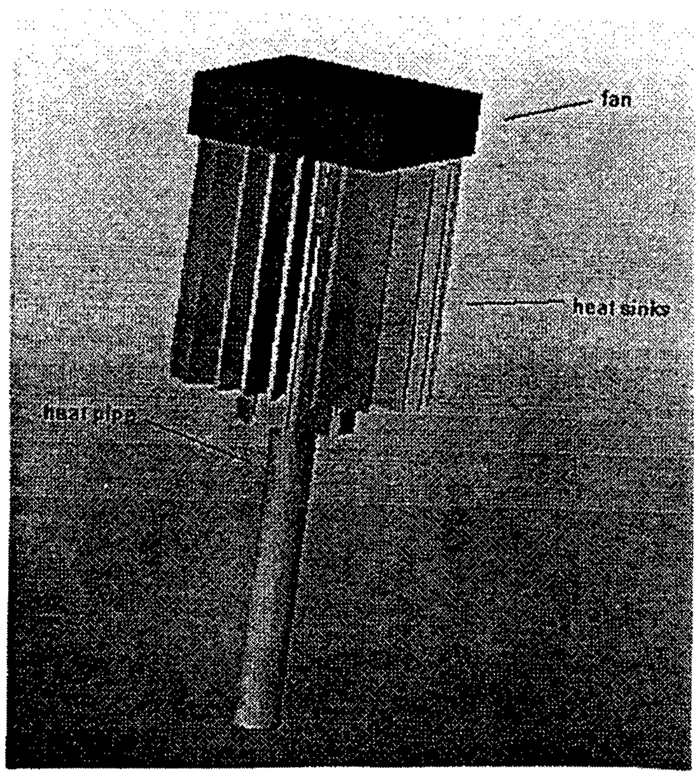

Figure 25. Second Generation Thermoelectric Module prototype Configuration (Front).

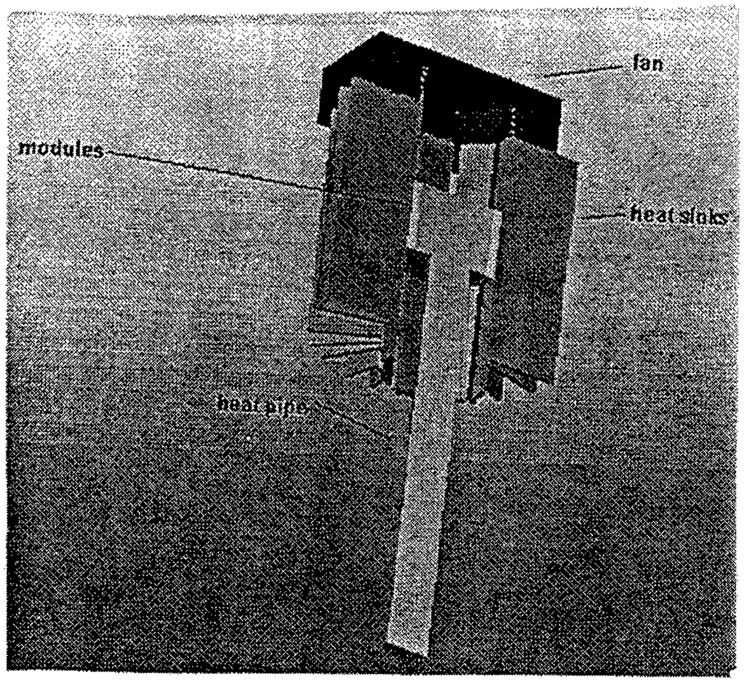

Figure 26. Second Generation Thermoelectric Module Prototype Configuration (Back).

This document reports research undertaken at the U.S. Army Soldier and Biological Chemical Command, Soldier Systems Center, Natick, MA, and has been assigned No. NATICK/TR-00/009 in a series of reports approved for publication.